ABSORPTION REFRIGERATION SYSTEM

by

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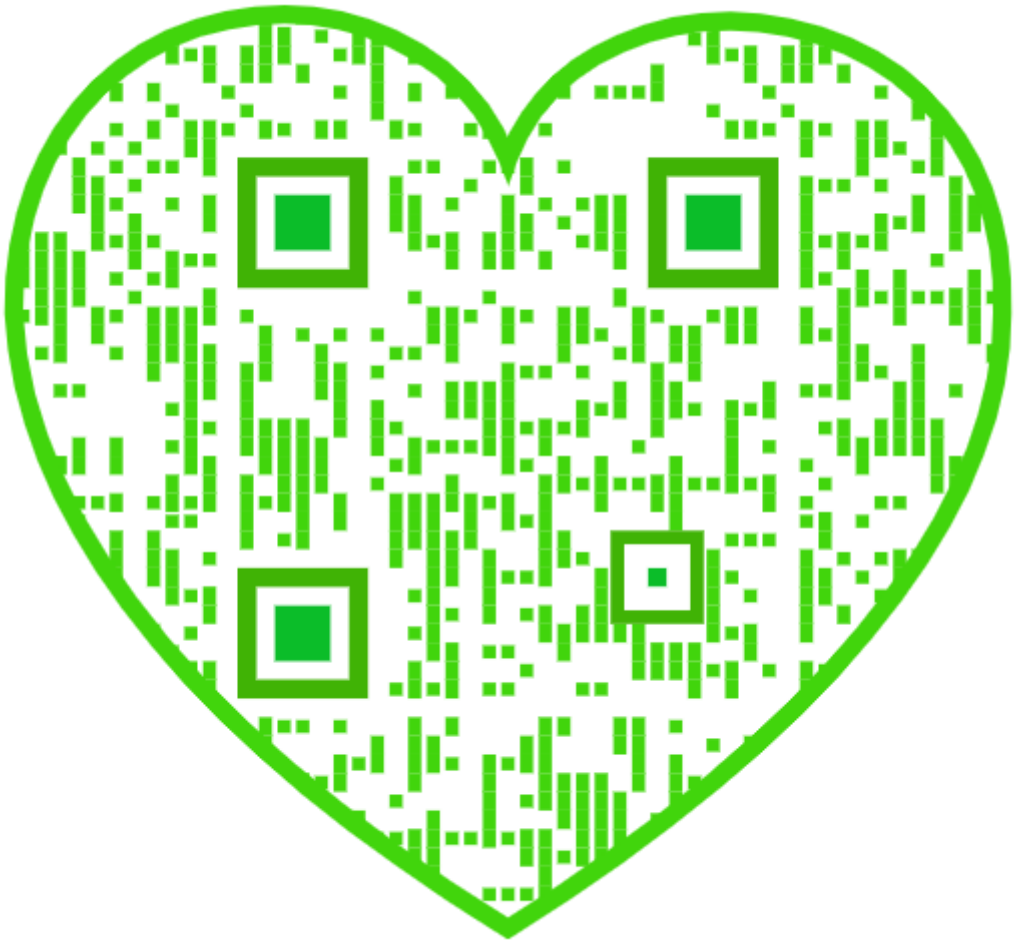

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SECTION 1

Introduction

1.1. Difference between Compression and Absorption Refrigeration Systems[1],[2]*

Refrigeration might be defined as the art of producing and maintaining, in a given space, a temperature level which is lower than the surrounding temperature level. The reversed Carnot cycle operating as a heat pump removes heat from a low temperature source and rejects it to a higher temperature region; no other cycle is more efficient than this one for given high temperature and low temperature sources. Actual refrigeration cycles, because of their inherent irreversibilities, operate at lower efficiencies than the reversed Carnot cycle. In the following discussions, irreversibility for each process in an actual cycle will be considered instead of the reversible processes which constitute the reversed Carnot cycle.

Generally speaking, refrigeration systems are divided into two classes —— compression refrigeration systems and absorption refrigeration systems. In a compression refrigeration system (see Fig. 1a), liquid refrigerant passes through a throttle valve, thereby undergoing a throttling process to a lower pressure and temperature. After expansion through the valve, the fluid is evaporated in the coils by means of the absorption of

* All superscripts with specified numbers in parentheses show bibliography which are tabulated on page 64.

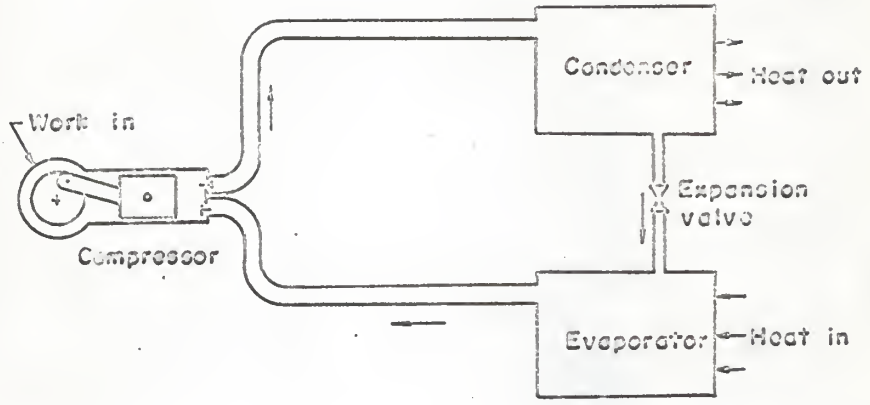

Fig. 1a. Basic compression refrigeration cycle

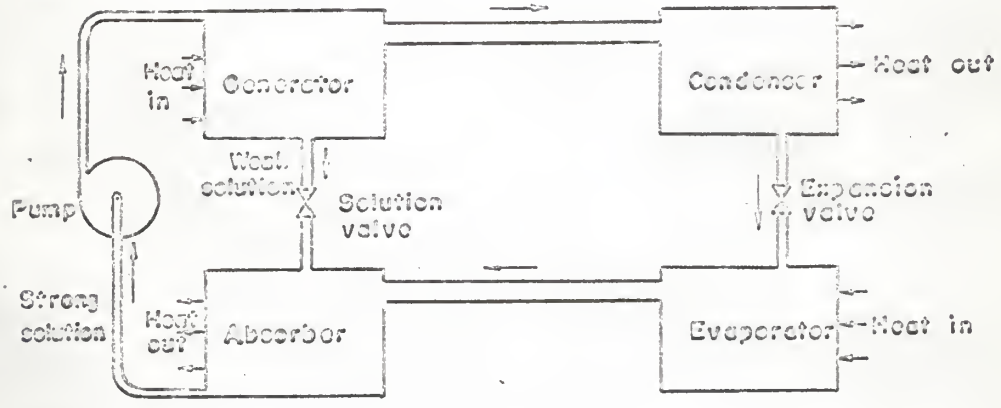

Fig. 1b. Basic absorption refrigeration cycle[2]

heat from the evaporator. The dry saturated vapor from the evaporator is then compressed to a higher pressure by supplying mechanical work to the compressor. During the compression the temperature of the refrigerant increases. The high temperature vapor is then condensed under constant pressure conditions to the entrance of the throttle valve. This completes the cycle.

Both compression refrigeration systems and absorption refrigeration systems have the same condensing process, throttling process through the expansion valve and evaporating process. The difference between the two systems is the process between the end of the evaporating process and the entrance of the condensing process. As mechanical work is much more expensive than an equivalent amount of heat, it may be desirable to use heat directly as the operating energy instead of using mechanical work. The most successful system operating almost wholly on an input of heat is the absorption system. The basic absorption refrigeration system (see Fig. 1b) uses an absorber, generator, solution valve and a liquid pump instead of the compressor in the compression refrigeration system and uses two fluids (such as ammonia and water) instead of one fluid (such as ammonia) in the compression refrigeration cycle. In accordance with the design pressure and temperature at the absorber, the mixed solution quickly reaches the equilibrium condition and finally becomes a strong concentrated solution. (A strong concentrated solution is one that has a relatively large amount of the refrigerant dissolved in the absorbing fluid.) During this mixing operation, cooling

water circulates continuously to take off this heat. The strong solution is then pumped to the generator where almost pure vapor flows to the condenser and the weak solution expands through a solution valve and flows back to the absorber. In the absorber the absorbent and the refrigerant vapor are brought into contact to facilitate the dissolving of vapor continuously. The basic difference described above can be seen by comparing the left hand sides of Fig. 1a and Fig. 1b. By far the most commonly used binary mixtures for absorption refrigeration systems are ammonia-water, lithium chloride-water, lithium bromide-water.

Of these two systems, the compression refrigeration system is much simpler in mechanism than the absorption refrigeration system. For comparisons, however, we have to consider some other practical and local circumstances, such as the investment cost, the availability of heating and cooling systems; we can not assert which system is preferable in general. Further detail in the comparison of the absorption and compression systems, by a purely thermodynamic viewpoint with the aid of performance ratio and coefficient of performance, will be shown in Sec. 2.3.

1.2. Principles of the Absorption Refrigeration System and Its Cyclic Analysis

Figure 2 shows a more complete diagram of a conventional absorption refrigeration system than does Fig. 1b. The high pressure vapor enters the condenser where it becomes a liquid. The liquid then passes through the expansion valve where it is throttled to a low pressure, low temperature, and low quality

vapor. This low pressure and low temperature refrigerant flows through the evaporator where it absorbs heat as the liquid is vaporized. The low pressure and low temperature vapor coming from the evaporator flows into the absorber where it comes in contact with the cool weak solution. The absorber operates at a pressure slightly lower than the evaporator pressure. The weak solution absorbs the vapor which comes from the evaporator and thereby becomes a strong solution. The maximum concentration of refrigerant vapor that can be absorbed depends upon the temperature and pressure in the absorber. This maximum absorption is an important concept in investigating absorption refrigeration systems. Fig. 3[4] shows, for example, the maximum composition of ammonia by weight in a solution of ammonia and water as a function of pressure and temperature.

During the absorption process, the heat of absorption, which is the equivalent of the heat of condensation of the liquid, must be removed for the purpose of holding a low temperature in the absorber and thereby maintaining a good absorption rate. Usually cooling water is used to absorb this heat.

From the absorber the strong solution* enters the pump at the system which raises its pressure and sends it through a heat exchanger and to the generator. The generator operates at the condenser pressure and is supplied with steam or other sources of

* "Rich solution" is also popularly used as an exchangeable term.

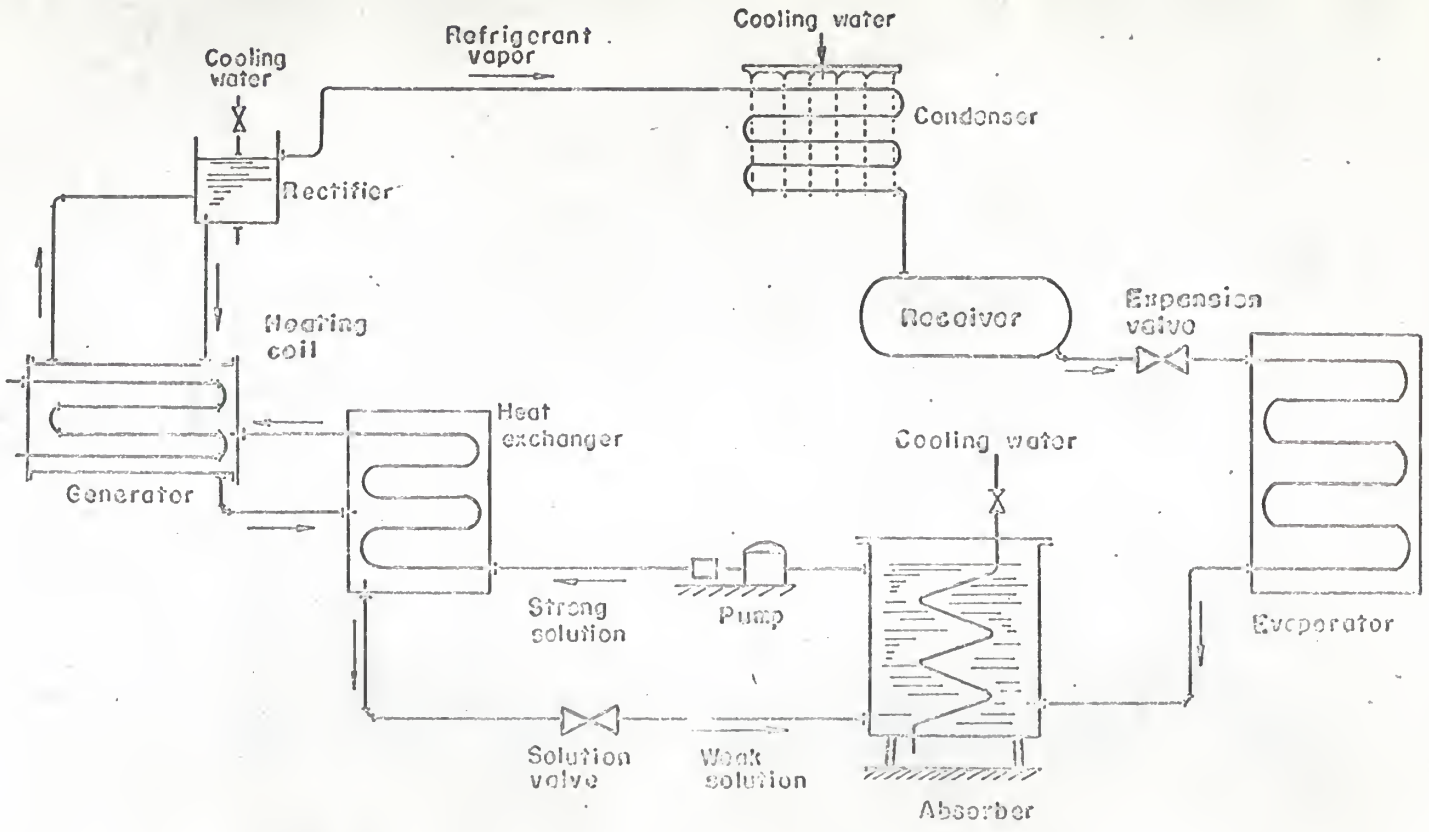

Fig. 2. Schematic diagram of an absorption refrigeration system

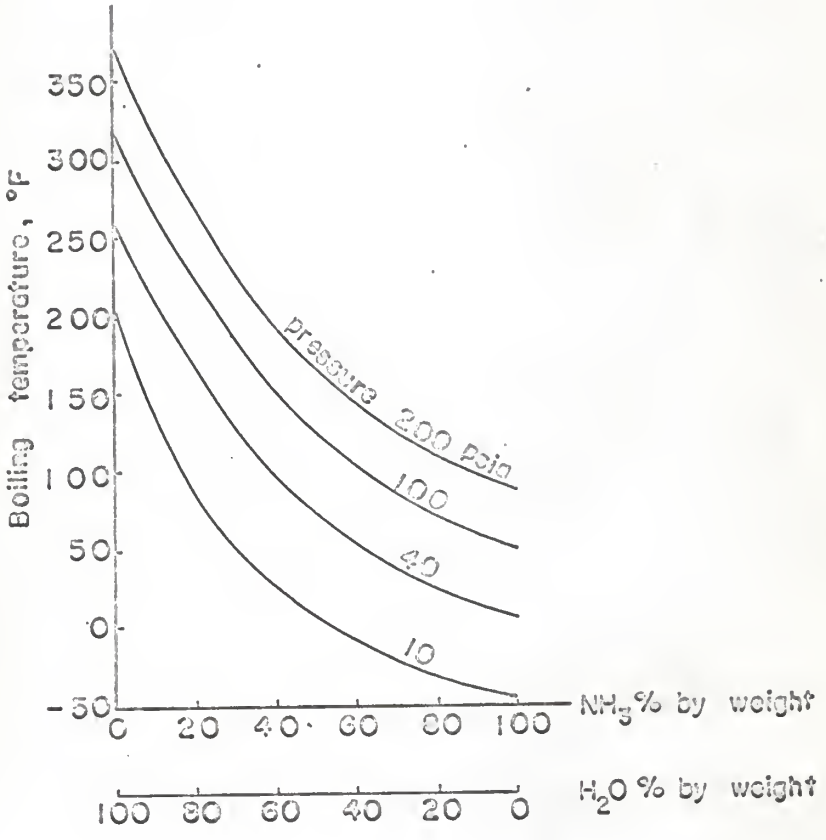

Fig. 3. Boiling temperature of aqua ammonia solution affected by pressure and concentration [4]

of heat* through heating coils which drives off some of the refrigerant from the liquid, decreasing the concentration of the liquid until it becomes a weak liquor. The vapor distilled from the solution in the generator is composed of refrigerant vapor along with small quantities of absorbent vapor, and when this mixture is cooled in the rectifier, the absorbent vapor saturated with refrigerant is condensed and flows back into the generator. By this means, part of the refrigerant is vaporized from the strong solution in the generator; the remaining solution is a mixture of liquid absorbent with a relatively small concentration of refrigerant dissolved in it. This remaining solution is called the weak solution. It leaves the generator, flows through the heat exchanger, the solution valve and then into the absorber.

In the heat exchanger the weak solution which comes from the generator heats the strong solution which is sent up from the absorber through the pump. This reduces the amount of heat required in the generator. Neglecting the pressure head losses due to friction, we can consider that in an absorption refrigerant system there are two distinct pressure levels which exist within the unit; the high pressure which exists in the heat exchanger, the generator, the rectifier and the condenser; the low pressure which exists in the absorber and the evaporator. After the weak solution has been cooled in the heat exchanger it enters the solution valve, which acts as a throttle valve, where its pres-

* Such as electrical heating; but steam heating is preferable due to its lower cost.

sure is reduced from the high pressure to the low pressure, and then flows into the absorber.

The refrigerant which was vaporized in the generator contains some absorbent vapor. Since this absorbent vapor will condense at a much lower temperature than the refrigerant vapor, this mixed solution is passed through a rectifier to condense the absorbent vapor in order to obtain pure refrigerant vapor. In the rectifier the refrigerant vapor is cooled sufficiently to condense the surplus absorbent which is separated and returned to the generator. It is important to keep the absorbent vapor to a minimum to prevent the accumulation of liquid or solid state absorbent in the condenser or refrigerant coils.

1.3. Cyclic Analysis of an Absorption Refrigeration System in Terms of the Circulating Fluids[5]

Attention has been previously directed to the principles of the absorption refrigeration system in terms of the processes that occur in each piece of apparatus. Now, the system will be analyzed from the viewpoint of the four sub-cycles which constitute the whole system.

(1) Refrigerant Vapor Cycle

After the strong solution in the generator is heated to the point of driving off some of the refrigerant vapor, with a small amount of absorbent, this vapor passes through the rectifier where the absorbent is condensed and the refrigerant cooled. The absorbent is separated and returned to

the generator while the refrigerant vapor passes through the condenser where it is condensed. The liquid then flows through the expansion valve to the evaporator where it takes up heat and is revaporized. From the evaporator it passes into the absorber and is absorbed by the cooled weak solution. This weak solution is thus changed to a strong solution which is pumped by the refrigerant pump through the heat exchanger to the generator. This completes the cycle.

(2) Weak Solution Cycle

When the strong solution is subjected to high temperature in the generator, some of the refrigerant in it is vaporized thus producing a weak solution. This weak solution is collected at the bottom of the generator.

In order to maintain constant liquid levels in the generator and the absorber, the flow of strong solution to the generator must be of such value that it equals the sum of the flows of weak solution from the generator and the refrigerant to the condenser. The weak solution flows from the generator to the heat exchanger where it is cooled. It then flows to the absorber where it absorbs the cold refrigerant coming from the evaporator, and is further cooled. The absorption changes the weak solution to a strong solution.

(3) Strong Solution Cycle

As mentioned in the refrigerant vapor cycle, the strong solution formed in the absorber by the union of refrigerant vapor and weak solution is picked up by the pump, forced through the heat exchanger where it is warmed, and then passes to the generator to be heated further.

(4) Cooling Water Cycle

When the cooling water is cold, it is common practice to let the water circulate first through the condenser. Here it liquefies the refrigerant and comes from the condenser sufficiently cold to be used in the absorber. From the absorber cooling water is circulated through the rectifier.

In warm weather, where a suitable supply of cold water is not available, separate water supplies are maintained for the condenser, the absorber and the rectifier.

Whether separate cooling water supplies are used or not, in either case the water coming from the cooling systems may be passed to the sewer discharge or carried either to cooling towers or spray ponds to be re-used. Whereas some power is used in sending such water through the cooling process of either spray ponds or cooling towers, there are certain advantages in reusing this water. The water used and reused does not scale so badly, since it has only so much hardness to deposit, and once this is separated the water is practically non-scaling. Further, unless very

cheap water is available, it is seldom possible to secure it, either by pumping or by purchase, at as low a cost as by cooling it either by sprays or cooling towers.

The four sub-cycles which have been described above can be seen from Fig. 4 on page 13.

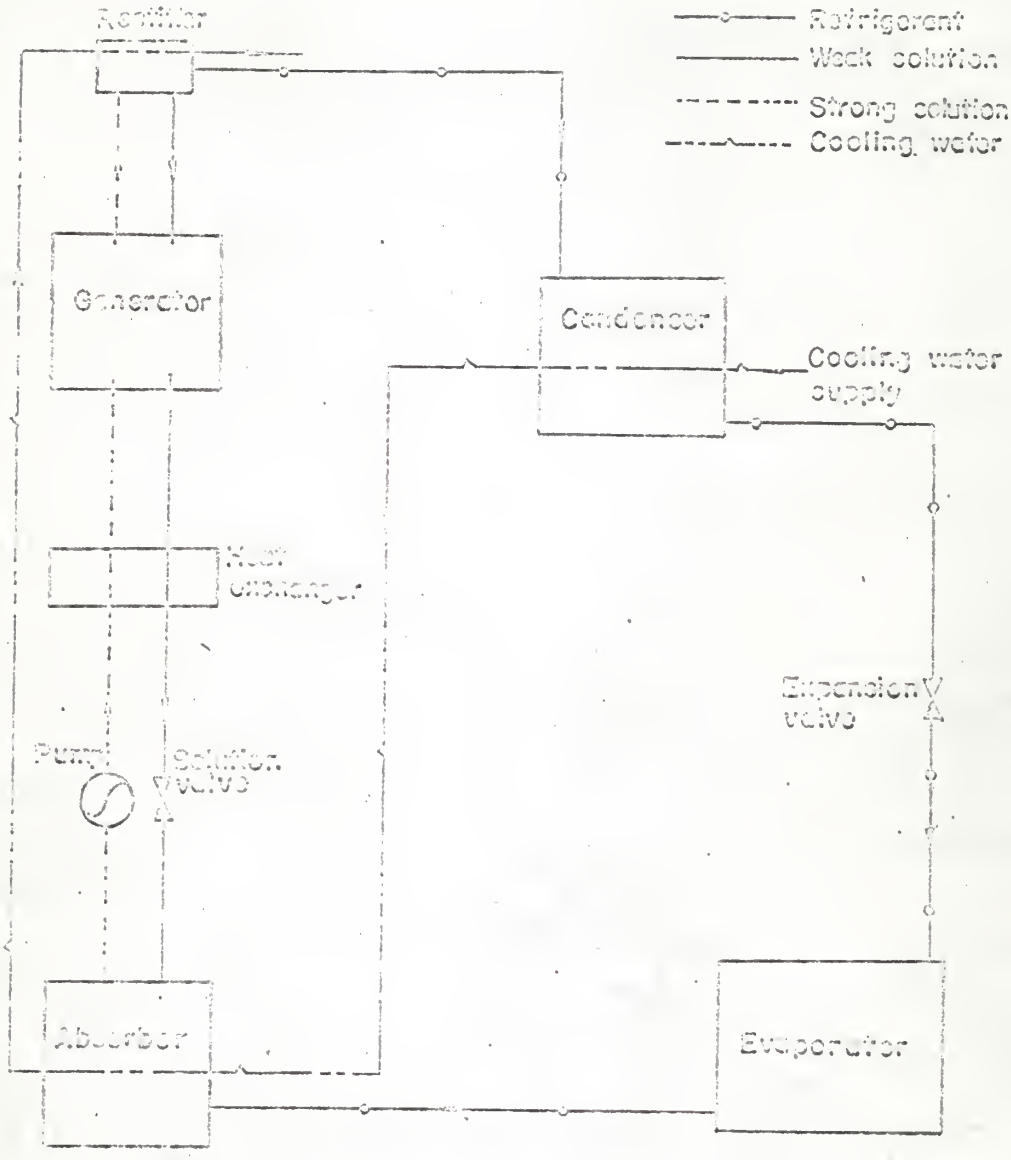

Fig. 4. Flow chart in terms of circulating fluids in an absorption refrigeration system

SECTION 2

First Law and Second Law Analysis of an Absorption Refrigeration System

Although the analysis of the reversible cycle by means of the reversed Carnot relation is useful for determining the limiting conditions, we shall now consider a more detailed analysis of a real cycle with irreversibilities. In this section the analysis consists of mass and energy balances around the various components as well as around the whole system.

A schematic diagram of an absorption refrigeration system is shown in Fig. 5(6). A binary mixture with appropriate properties (for example, NH_3 - H_2O, H_2So_4 - H_2O) is vaporized in the generator by the addition of a quantity of heat Q_g at the generator pressure P_H. As a result vapor refrigerant, for example, ammonia, is produced. Let x = 0 denote pure absorbent and x = 1 denote pure refrigerant concentration. The pressure P_H is chosen so that the entire refrigerant vapor can be liquefied in the condenser. For the fluid flow through the expansion valve, the condensate is throttled to the lower pressure P_L, and the saturation temperature falls. In this way the liquid can evaporate at a lower vaporization temperature, T_e, with the addition of the heat quantity Q_e (the so-called refrigeration capacity).

The cool refrigerant vapor (3)* coming from the evaporator

* Specified numbers in parentheses, ( ), refer to the corresponding sections which is labeled in Fig. 5.

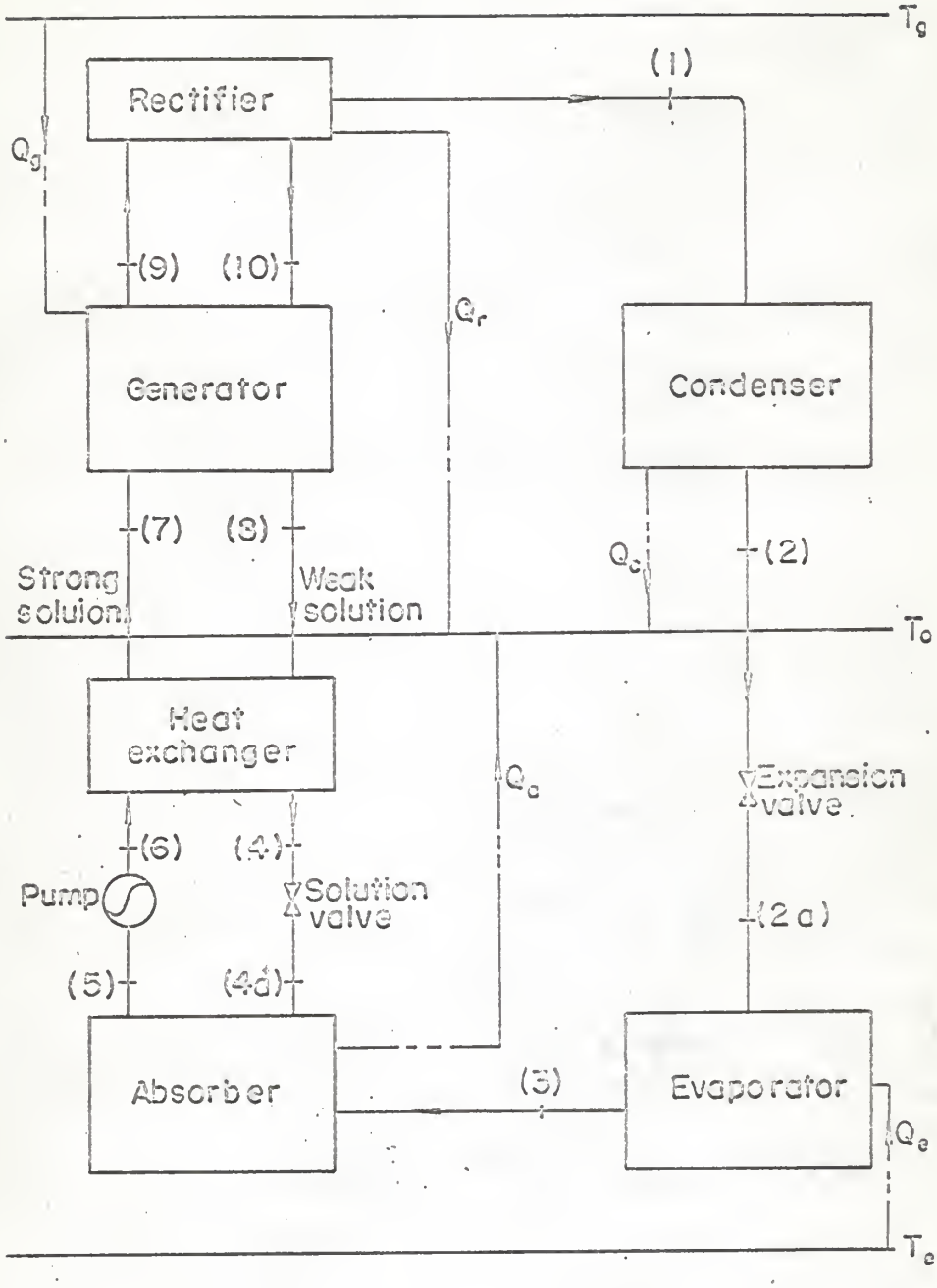

Fig.5. Typical absorption refrigeration system[6]

is sent into the absorber where it is absorbed by the cool solution with the removal of an amount of heat Q_a. The refrigeration process of the absorption system is based on the fact that a cold vapor can be absorbed by a warm liquid solution.

The weak solution of composition, x_w, comes from the generator, passes through the heat exchanger, is throttled in the solution valve to the absorber pressure, P_L, and passes into the absorber. The cold vapor of state (3) is induced, and this is absorbed by the weak solution, thereby becoming a strong solution. The absorption of vapor produces heat. In order to maintain the absorber at a sufficiently low temperature, cooling water is circulated through coils in the absorber. The enriched cold solution is pumped to the higher generating pressure, P_H, warmed through the heat exchanger, and is again vaproized in the generator. The rectifier provides the final drying effect by cooling and delivering pure refrigerant vapor to the condenser. The drip from the rectifier is returned to the generator. In this way the total cycle continuously works without interruption.

2.1. Mass Balance[6],[7]

In steady-state operating conditions, the quantities of mass supplied to and removed from each apparatus must be equal. This refers both to the total quantity of the mixture as well as to a simple constituent.

For the purpose of finding the rate of flow of pure refrigerant to the condenser, let M_n, where n = 1, 2, 2a, 3, 4, 4a, 5, ..., 10, denote the mass of fluid that crosses the various

sections indicated on Fig. 5.

Hence, at the absorber, the refrigerant mass balance is

$$M_5 x_5 = (M_5 - M_3)x_w + M_3 x_3 \qquad (1)$$

in which $x_s = x_5 = x_6 = x_7$, denotes the weight concentrations of the strong liquid solution, and

$$x_w = x_4 = x_{4a} = x_8 = x_{10},$$

denotes weight concentration of the weak solution.

For each pound of refrigerant absorbed from the evaporator, the amount of refrigerant circulated in terms of the weight concentrations of strong solution (x_s) and of weak solution (x_w) is

$$G = \frac{M_5}{M_3} = \frac{1 - x_w}{x_s - x_w} \qquad (lb_m/lb_m \text{ of } M_3) \qquad (2)$$

This means that for each pound of the refrigerant vapor produced, the specific solution quantity G must be pumped from the absorber through the heat exchanger to the generator.

The quantity of weak solution that flows back to the absorber is

$$G\text{-}1 = \frac{M_5 - M_3}{M_3} = \frac{1 - x_s}{x_s - x_w} \qquad (lb_m/lb_m \text{ of } M_3). \qquad (3)$$

From equations (2) and (3) we can see that the smaller the degassing breadth, $x_s - x_w$, the larger must be the specific solution circulated; nevertheless, this is still restricted by the chemical composition of the refrigerant absorbent solution.

Further

$$M_1 = M_2 = M_{2a} = M_3 \qquad (4)$$

$$M_5 = M_6 = M_7 = M_8 + 1 \qquad (4)$$

In assuming the fluid flows steadily, the amounts of material stored in the generator and the rectifier do not change with time. For the refrigerant flow in the generator,

$$x_7 M_7 = x_8 M_8 + 1 \qquad (5)$$

Substituting the assumed values for x_7 and x_8, or x_s and x_w, into (5) we can get the values of M_7 and M_8.

To find the amount of vapor leaving the generator and the amount of condensate returning to it, we equate the quantities flowing into and out of the rectifier; we have

$$M_9 = M_{10} + 1 \qquad (6)$$

For the refrigerant alone, we have

$$x_9 M_9 = x_{10} M_{10} + 1 \qquad (7)$$

where x_{10}, as has been shown above, is identical with x_8. The concentration x_9, on the other hand, is that for the refrigerant and absorbent in the vapor phase which is in equilibrium with the weak solution (liquid phase) at the pressure and temperature existing in the generator. To find the maximum values of x_9 and x_{10}, we can refer to tables and charts which give the maximum concentrations of the refrigerant and absorbent, for both the vapor and liquid phases, as functions of the pressure and temperature.

2.2. Energy Balance[5],[6]

For any piece of apparatus, the energy equation for steady flow, neglecting the change in kinetic energy and potential

energy terms, is

$$E = \Sigma(Mh)_{out} - \Sigma(Mh)_{in} \tag{8}$$

where E denotes the work input for the pump or heat transfer for all other pieces of equipments, $\Sigma(Mh)_{out}$ denotes the enthalpy of all the fluids that flow out while $\Sigma(Mh)_{in}$ denotes the enthalpy for all the fluids that flow in, based on unit mass of the refrigerant flowing through the evaporator.

(1) Heat Exchanger

Neglecting heat transfer with the surroundings, the equation is

$$M_7h_7 + M_4h_4 - M_6h_6 - M_8h_8 = 0 \tag{9}$$

(2) Generator

In the generator, the energy input of the strong solution is M_7h_7, the energy supplied by the drips from the rectifier is $M_{10}h_{10}$, the energy leaving with the weak solution, as it flows into the heat exchanger, is M_8h_8, and the energy leaving with the vapor as it flows to the rectifier is M_9h_9. Hence the energy balance is

$$E = Q_g = M_8h_8 + M_9h_9 - M_7h_7 - M_{10}h_{10} \tag{10}$$

where Q_g denotes the amount of heat supplied to the generator per unit mass of refrigerant passing through the evaporator.

(3) Similarly, for the rectifier, the condenser, the evaporator and the absorber, we can get the simplified energy equations according to equation (8) as follows:

$$Q_r = h_1 + M_{10}h_{10} - M_9h_9 \tag{11}$$

$$Q_c = h_2 - h_1 \tag{12}$$

$$Q_e = h_3 - h_{2a} \tag{13}$$

$$Q_a = M_5h_5 - M_{4a}h_{4a} - h_3 \tag{14}$$

where Q_r, Q_c, Q_e and Q_a represent the amount of net heat transfer in the rectifier, the condenser, the evaporator and the absorber, respectively.

(4) Pump Work

This can be calculated approximately by the usual equation for ideal pump work

$$Q_p = \frac{144\ G\ V_s(P_M - P_L)}{778} \tag{15}$$

in which Q_p denotes the ideal pump work input in BTU per pound of refrigerant vapor which circulates through the evaporator; P_H and P_L denote the generator and the evaporator pressure in pounds per square inch absolute, respectively; V_s denotes the specific volume in cubic feet per pound of strong solution as it enters the pump from the absorber.

Or, by equation (8)

$$Q_p = M_6h_6 - M_5h_5 = M_6(h_6 - h_5) \tag{16}$$

(5) Overall Heat Balance

For a heat balance, all energies supplied to and removed from the whole system are identical. Summarizing the above energy equations, we have

$$Q_a + Q_c + Q_r + Q_g + Q_p + Q_e = 0 \qquad (17)$$

(6) Steam Consumption

A commercial ton of refrigeration is equal to the absorption of heat in the evaporator at the rate of 288,000 BTU per day or 12,000 BTU per hour. The value of 12,000 BTU per hour is based on the heat required to melt one ton of ice in one day or $144 \times \frac{2000}{24} = 12,000$ BTU/hour, where 144 BTU is the latent heat of fusion of ice at atmospheric pressure. Thus the steam consumption in the generator per ton per hour is

$$\frac{Q_g}{Q_e} \cdot \frac{12,000}{h_{fg}} \qquad (18)$$

where h_{fg} is the latent heat of the heating medium (such as steam) at the temperature of the heating coil.

2.3. Second Law Analysis of An Absorption Refrigeration System

(1) Introduction

The Second Law of thermodynamics states that no process can be devised whose sole result is the absorption of heat from a reservoir (source) of one temperature and transfer of this heat to a reservoir of higher temperature; or by itself

heat can not flow, by any means, uphill. In accordance with this statement, for the purpose of the transfer of a certain amount of heat from a cold storage temperature to the surrounding temperature, there must exist a process which can cause this heat transfer; however, the Second Law of thermodynamics has not restricted the kinds of necessary compensating process. Therefore, the process may be selected as mechanical work of compression, or by thermal energy input combined with fluids that have suitable properties. The absorption refrigeration system is considered to be a successful system in utilizing the combination of the recorded methods.

Furthermore, there are a number of additional ways by which it might be possible to effect varying degrees of improvement in the performance of an absorption refrigeration system. To discuss all such methods in detail is beyond the scope of this report. Before we consider the problem to improve the performance of an absorption refrigeration system we have to have a concept of determining the minimum heat expenditure required to attain a desired cooling capacity.

In this section, entropy change for each process, performance ratio, concepts of availability and irreversibility are involved.

(2) Entropy Changes

No cycle is more efficient than a completely reversible cycle. If each process of a cycle undergoes a revers-

ible process, and all other systems associated with the cycle undergo reversible process, the total change of entropy of the universe is zero. Nevertheless, if any process of a cycle undergoes an irreversible process, the entropy of the universe must be increased due to the entropy creation during this process. This also means that the amount of entropy change due to the irreversibility is the direct measurement of the resulting loss on efficiency due to the irreversible process.

Since entropy is a state property, the entropy change of any process, regardless of the reversibility or irreversibility of the process, can be evaluated by the integral of $\frac{dQ}{T}$ along any reversible path, or by the equational form

$$S_{II} - S_I = \int_I^{II} \left(\frac{dQ}{T}\right)_{rev} \tag{19}$$

in which subscripts I and II denote initial and final state of a process, respectively.

For a cyclic analysis, the end state being the same as the initial state, $\oint ds = 0$.

Furthermore, by the Clausius inequality

$$\oint \frac{dQ}{T} \leq 0 \tag{20}$$

where the equal sign indicates the limiting or reversible cycle, while $\oint \frac{dQ}{T} < 0$ holds for any cycle which is composed of irreversible processes.

The general form of the entropy balance equations, for any process, can be expressed by

$$\Sigma S_{in} + S_{created} + \int \frac{dQ}{T} = \Sigma S_{out} + \Delta S_{stored} \qquad (21)$$

in which ΣS_{in} and ΣS_{out} are entropy fluxes due to the fluid flow, and T is the temperature of the fluid which emits the heat which is transferred.

In the following discussions, in which the system is in steady state condition, the last term of the above equation is zero. Further, in an analysis of an absorption refrigeration system, as has been shown diagramatically on Fig. 5, we shall consider the surroundings, or atmosphere, as the cooling water that flows to the condenser. The temperature of this cooling water (or atmosphere) is designated as T_0. The temperature of the steam supplying heat to the generator and the refrigerated space are each assumed to be constant and are designated as T_g and T_e, respectively.

Further, in an analysis of a practical problem, a finite temperature difference must exist in each heating or cooling process, so that the heat can be transferred. Nevertheless, if we assume that in the absorber there is a great amount of cooling water which circulates through the absorber, the temperature of the absorber can be taken as the cooling water inlet temperature instead of taking the average temperature of the cooling water in the absorber. This will be done also for the condenser and for the rectifier. The total heat flow to the surrounding, or cooling water at temperature T_0, is composed of the heat transferred from the absorber (Q_a), that transferred from the condenser

(Q_c), and the rest from the rectifier (Q_r). Consequently, the total amount of entropy increase of the cooling water is

$$S_{a+c+r} = -\frac{Q_a + Q_c + Q_r}{T_0} \tag{22}$$

Equations of entropy balances for these three processes, according to the equation $\Sigma S_{created} = \Sigma S_{out} - \Sigma S_{in} - \int \frac{dQ}{T}$ under steady state conditions, are, for each pound of refrigerant condensed,

$$S_{created,a} = (M_5 S_5 - M_{4a} S_{4a} - S_3) - \int \frac{dQ_a}{T_0} \tag{23}$$

$$S_{created,c} = (S_2 - S_1) - \int \frac{dQ_c}{T_c} \tag{24}$$

$$S_{created,r} = (S_1 + M_{10} S_{10} - M_9 S_9) - \int \frac{dQ_r}{T_r} \tag{25}$$

$S_{created,a}$ represents the amount of entropy creation due to the mixing of fluids of different temperatures in the absorber. In the condenser, if the pressure drops due to fluid friction, and a finite temperature difference is considered to exist between the condensing refrigerant and the cooling water, the total amount of entropy creation due to these two irreversibilities is

$$S'_{created,c} = \Delta S_{f.c} + \Delta S_{\Delta T_c}$$

$$= \Delta S_{f.c} + \left(\frac{Q_c}{T_0} - \frac{Q_c}{T_c}\right) \tag{26}$$

where subscripts f and ΔT_c denote the friction and the finite temperature difference, $(T_c - T_0)$, in the condenser,

respectively. This method of analysis is still available for other apparatus, such as the evaporator, the generator, or the absorber, if the finite temperature differences are not neglected. In the evaporator, in the present analysis, we still assume that the amount of heat received by the refrigerant in the evaporating coil is transferred from a large reservoir at the saturation temperature of the refrigerant in the evaporator so that the finite temperature difference is neglected. The amount of entropy flow from the refrigerated space to the evaporating refrigerant is $\int \frac{dQ_e}{T_e}$. Hence the equation of entropy balance for the evaporating process becomes

$$S_{created,e} = (S_{2a} - S_2) - \int \frac{dQ_e}{T_e} = 0 \qquad (27)$$

Similarly, for the generating process,

$$S_{created,g} = (M_8S_8 + M_9S_9 - M_7S_7 - M_{10}S_{10}) - \int \frac{dQ_g}{T_g} \qquad (28)$$

The pumping process is considered here to be an isentropic process, i.e.

$$S_{created,p} = 0 \qquad (29)$$

For the heat exchanger, the solution valve, and the expansion valve, we assume that there are no heat transfers to or from surroundings, i.e. $Q_{ex} = Q_{vs} = Q_{re} = 0$. Hence, the equations of entropy balances for these three apparatus are

$$S_{created,ex.} = M_6(S_7 - S_6) + M_4(S_4 - S_8) \quad (30)$$

$$S_{created,vs.} = M_4(S_{4a} - S_4) \quad (31)$$

$$S_{created,ve.} = S_{2a} - S_2 \quad (32)$$

The Clausius inequality for the non-ideal cycle is

$$\oint \frac{dQ}{T} = \frac{Q_g}{T_g} + \frac{Q_e}{T_e} + \frac{Q_a}{T_a} + \frac{Q_r}{T_r} + \frac{Q_c}{T_c} < 0 \quad (33)$$

and for each process

$$S_{II} - S_I \geq \int_I^{II} \left(\frac{dQ}{T}\right) \quad (34)$$

Further,

$$\Delta S_{universe} = \Sigma S_{created}$$

$$= \Delta S_{atmosphere} - (S_e + S_g) \quad (35)$$

in which

$$\Delta S_{atmosphere} = S_e + S_g + S_{created,g} + S_{created,a} + S_{created,c} + S_{created,r} + S_{created,e} + S_{created,ex} + S_{created,vs} + S_{created,ve} \quad (36)$$

(3) Performance Ratio*

The amount of work and/or heat required to move a certain quantity of heat from the cold storage room is an important factor in the analysis of an refrigeration system. In a compression refrigeration system, this factor is expressed by "coefficient of performance"; however, in an

* Some authors use "heat ratio" as an exchangeable term.

absorption refrigeration system the total energy input is the sum of the mechanical work which is done by the solution pump and the thermal energy which is supplied from the heat source temperature, T_g. Therefore, before considering the performance ratio, we have to consider, with the aid of the Second Law, how much equivalent quantity of heat is considered to be supplied from the heat source temperature, T_g, corresponding to the actual pump work input. This equivalent amount of heat may be expressed by the relation

$$Q_{pp} = \frac{T_g}{T_g - T_0} Q_p \tag{37}$$

From equations (17) and (33), we have

$$\frac{Q_g}{T_g} + \frac{Q_e}{T_e} \leq \frac{Q_a + Q_e + Q_r}{T_0} \tag{38}$$

Substituting equation (17) into (38), and after rearranging, we get

$$Q_g + \frac{T_g}{T_g - T_0} Q_p \geq \frac{T_0 - T_e}{T_e} \cdot \frac{T_g}{T_g - T_0} \tag{39}$$

The performance ratio, Φ_α^*, of an absorption refrigeration system is defined[10] as

$$\Phi_\alpha = \frac{Q_e}{Q_g + Q_{pp}} = \frac{Q_e}{Q_g + \frac{T_g}{T_g - T_0} Q_p} \tag{40}$$

* In this section the subscripts α and k denote absorption refrigeration systems and compression refrigeration systems, respectively.

Substituting equation (39) into the denominator of equation (40), we get

$$\Phi_\alpha = \frac{Q_e}{Q_g + \frac{T_p}{T_g - T_0} Q_p} = \frac{Q_e}{Q_e \frac{T_0 - T_e}{T_e} \cdot \frac{T_g}{T_g - T_0}}$$

$$= \frac{T_e}{T_0 - T_e} \cdot \frac{T_g - T_0}{T_g} = (CP)_{rev} (\eta)_{rev} \quad (41)$$

where

$$(CP)_{rev} = \frac{T_e}{T_0 - T_e} \quad (42)$$

is the coefficient of performance of a refrigeration process of a reversed Carnot cycle (with mechanical drive) between the surrounding temperature, T_0, and the refrigerated space temperature, T_e, and

$$(\eta)_{rev} = \frac{T_g - T_0}{T_g} \quad (43)$$

is the thermal efficiency of a Carnot engine which operates between the heat source temperature, T_g, and the surrounding temperature, T_0. If the energy is assumed to be supplied by the quantity of equivalent heat, $(Q_g + Q_{pp})$, the performance ratio, Φ_α, of the absorption refrigeration system will then be the limiting case of a reversible cycle; that is

$$\Phi_{\alpha,max.} = (CP)_{rev} (\eta)_{rev} \quad (44)$$

This is the combined, ideal case of a reversible refrigerator operating between the surrounding temperature, T_0, and the refrigerated space temperature, T_e, with coefficient of

performance, $(CP)_{rev}$, and a reversible heat engine operating between the heat source temperature, T_g, and the surrounding temperature, T_0, which is supplied heat by the amount $(Q_g + Q_{pp})$, and which operates with the thermal efficiency $(\eta)_{rev}$. In an actual case, where irreversibilities exist, the actual performance ratio, Φ_α, of an absorption refrigeration system must be less than the ideal, or maximum, value, $\Phi_{\alpha,max}$. Or, expressing this in equational form, we have

$$\Phi_\alpha < \Phi_{\alpha,max}. \tag{45}$$

Furthermore, the value of $\Phi_{\alpha,max}$. (which is the reversible case) depends upon the temperatures T_g, T_0 and T_e; its value can be greater than or smaller than unity. However, the performance ratio, Φ_α, gives a concept for measuring the fraction of energy input $(Q_g + Q_{pp})$ which can be converted into the refrigerating capacity Q_e. The quotient

$$\epsilon = \frac{\Phi_\alpha}{\Phi_{\alpha,max}} \tag{46}$$

gives the efficiency ratio of the process in which $0 < \Phi_\alpha < 1$. The difference

$$\mu = 1 - \epsilon \tag{47}$$

denotes the degree of irreversibility, which shows that as a result of irreversibility the μ-th part of the input energy $(Q_g + Q_{pp})$ is degraded.

Obviously, we can not directly compare the performance ratio, Φ_α, of an absorption refrigeration system with the

coefficient of performance, $(CP)_k$, of a compression refrigeration system (equal to the ratio, $JQ_e/W_{comp.}$) even if these two different types of refrigerators were assumed to be operated between the same temperature range of T_0 and T_e. However, a comparison can be made between $(CP)_\alpha$ (the coefficient of performance of the absorption cycle) and $(CP)_k$.

Consider a thermodynamic efficiency term, η_t, defined as the ratio of the sum of the increase in available energies of the fluid in the generator and in the pump, and the sum of the heat terms, Q_g and $\frac{T_g}{T_g - T_0} Q_p$. For the ideal case, in which there is no degradation of energy in the generator or the pump, $\eta_t = \frac{T_g - T_0}{T_g}$; in the actual case $\eta_t < \eta_{t,max}$. The coefficient of performance is defined

$$(CP)_\alpha = \frac{\Phi_\alpha}{\eta_t} = \frac{Q_e}{(Q_g + \frac{T_g}{T_g - T_0} Q_p)\eta_t} \qquad (48)$$

while, as given above

$$(CP)_k = \frac{JQ_e}{W_{comp.}}$$

Further details will be shown numerically in Sec. 3.2.

(4) Concepts of Availability and Irreversibility[6],[11],[13]

In basic analyses for problems of thermodynamics, it is often maintained that thermodynamics is concerned with reversible processes and equilibrium states, and that problems associated with irreversibility lie outside its scope. On the other hand, the First Law of thermodynamics states that

heat and work are mutually convertible, but, since energy can neither be created nor destroyed, the total energy associated with an energy conversion remains constant. However, the thermodynamic analysis of a system exchanging heat and work with its surroundings can not be more than the tabulation of energies received and rejected by the system when the analysis is based solely on the First Law of thermodynamics. However, the Second Law analysis, on the other hand, is based on the concept of available and unavailable energy associated with the degradation of available energy by irreversibilities.

High grade energy may be transformed into the mechanical work with only small losses. The losses of high grade energy are due to the mechanical or electrical imperfections within the devices themselves, but not because of some theoretical limitation of energy transformation. For instance, of the total amount of rotary shaft work which is delivered to a pump or compressor, a large fraction of it is delivered to the fluid as mechanical energy with only a minor fraction of it degraded by friction. Of lower grade energy, such as thermal energy, only some part of it (the available energy) can be transformed into the useful work while the rest of it is unavailable energy which can not be used as rotary shaft work. Suppose that there is a state of a system which is not at the dead state, then it will naturally and simultaneously change its conditions towards

be delivered by the fluid to the surroundings is the quantity given by a reversible engine which brings the fluid from state (I) to the state (D) which is in equilibrium with the surroundings.

The maximum amount of work is established by assuming that the heat given out by the fluid is delivered in an indefinitely large number of steps to a series of Carnot engines which work between the temperature of the fluid at any instant as source and the temperature of the surroundings. The work delivered by the engines may be considered to be stored as mechanical energy in external systems. Assuming the fluid has reached a certain state with pressure P, temperature T, and volume V, then the abstraction of dq units of heat from the fluid delivers $(-\frac{T - T_D}{T} dq)$ units of work from the Carnot engine operating at this temperature level. The quantity $(-\frac{T - T_D}{T} dq)$, termed "the motivity" by Kelvin, is always greater than zero because the terms $(T_D - T)$ and dq are always of the same sign; for instance, dq is positive when heat is received by the fluid and $(T_D - T)$, or $-(T - T_D)$, is positive. The reverse case is also true when the heat is given up by the fluid. Other than the work obtained from the exchange of energy between the fluid and surroundings, work may be obtained further if a pressure difference exists, i.e. if $P \neq P_D$. Hence the maximum work which is obtainable in the step is

$$dW_{max} = \frac{T_D - T}{T} dq + (P - P_D)dV \tag{50}$$

From the First Law, we have

$$dq = dU + PdV \tag{51}$$

and for a reversible process, we have

$$dS = \left(\frac{dq}{T}\right)_{rev.} \tag{52}$$

Solving equations (50), (51) and (52), we get

$$dW_{max} = - dU - P_D dV + T_D dS \tag{53}$$

Integrating from state (I) to state (D) gives

$$W_{max} = (U_I - U_D) - P_D(V_I - V_D) - T_D(S_I - S_D)$$

or

$$W_{max} = (U_I + P_D V_I - T_D S_I) - (U_D + P_D V_D - T_D S_D) \tag{54}$$

The value of the difference of equation (54) is called the available energy at state (I). For any two states, say (I) and (II), the difference of available energy with respect to the surrounding temperature and pressure of T_D and P_D is given by [13]

$$\Delta AE = (U_{II} + P_D V_{II} - T_D S_{II}) - (U_I + P_D V_I - T_D S_I) \tag{55}$$

Keenan[6] expresses, the maximum work obtainable as represented by the difference in the availability to produce work between the initial state and dead state, designated as the decrease of availability ($-\Delta B$) in the following expression.

$$W_{max} = -\Delta B = B \tag{56}$$

The available part of flow work is added to equation

(54) resulting in

$$\left[U - T_D S + P_D V\right]_D^I + (P_I - P_D)V_I$$

$$= \left[U - T_D S + P_D V + (P - P_D)V \right]_D^I$$

$$= \left[U - T_D S + PV \right]_D^I$$

$$= \left[h - T_D S \right]_D^I \qquad (57)$$

Accordingly, the change of availability between any two states, (I) and (II), in a steady flow process can be evaluated by

$$\Delta b = b_{II} - b_I = (h_{II} - T_D S_{II}) - (h_I - T_D S_I) \qquad (58)$$

In general, changes of kinetic energy and potential energy terms should be considered; however, for analyzing refrigeration systems, these terms are all small compared with the total energy at each section and are reasonably considered to be negligible.

As multiple streams enter and leave an apparatus, such as the absorber, the generator, the heat exchanger, or the rectifier, in an absorption refrigeration system, equation (58) has the more general form as follows,

$$\Delta B = \sum_{II=1}^{n} M_{II} b_{II} - \sum_{I=1}^{m} M_I B_I \qquad (59)$$

in which M denotes mass; I and II denote inlet and outlet conditions respectively; m and n denote the number of inlet

and outlet streams, respectively.

From equation (56) it can be seen that for any reversible process, the decrease of availability equals the maximum amount of work done by the system while the maximum increment of availability equals the work added to the system. For any irreversible process

$$W_{irr.} < W_{max}$$

therefore

$$W_{irr.} < -\Delta B \tag{60}$$

Hence, it is possible to express the irreversibility of any work producing process by the difference between the decrease of availability and the actual work obtained.

The irreversibility (I) can then be evaluated by the equation

$$I = \int \frac{T - T_D}{T} dQ - W_s - \Delta B \tag{61}$$

where $\int \frac{T - T_D}{T} dQ$ is the available part of the heat added during the process, W_s is the rotary shaft work output during the process and ΔB is the change in availability of the system during the process.

Furthermore, by the Clausius inequality

$$\int dS \geq \int \frac{dQ}{T}$$

or

$$\int T_0 dS \geq T_0 \int \frac{dQ}{T}$$

For each process the irreversibility can also be expressed

by

$$I = \int T_0 dS - T_0 \int \frac{dQ}{T}$$

$$= \Sigma(T_0 \Delta S) - T_0 \int \frac{dQ}{T} .$$

For any closed cycle

$$\Sigma(T_0 \Delta S) = 0$$

Therefore, the irreversibility for the whole cycle can also be expressed by

$$I = - T_0 \Sigma \frac{dQ}{T} \qquad (62)$$

SECTION 3

Computation of a Typical Ammonia-Absorption Refrigeration System

3.1. Principal Determining Factors

The complete cycle of operation of the absorption refrigerating machine is illustrated with the aid of diagram in the previous sections. Before proceeding with the numerical problem for a complete cycle, it is well to note what are the determining factors which are assumeable before working the particular problem.

The condenser operates in the same manner as in the compression system. The temperature of the cooling water, in the most part, determines the condenser pressure. However, the quantity of water and amount of condenser surface must be of sufficient magnitude to realize a desirable condenser pressure. The temperature of the water rises a few degrees in passing through the condenser, and there is a small temperature difference between the water leaving the condenser and the temperature of the condensing refrigerant in the saturated portion of the condenser.

The temperature of the evaporating fluid in the evaporator is determined by the temperature desired in the cold storage room. The difference in the temperature of the cold storage room and the evaporating fluid is also affected by the amount of heat transfer surface which is used. The temperature of the evaporator must always be a few degrees below the temperature of the

cold storage room. The pressures, of course, correspond to the various boiling temperatures of the fluid in the evaporator.

In a like manner, the temperature of the water available for cooling the absorber affects the conditions in the absorber. It is evident that the absorber must be supplied with cooling water, since the changes of states, such as condensation, are accompanied by a liberation of heat. With consideration to the relative amount of heat transmitting surface in the absorber, it is evident that the temperature of the cooling water will determine the temperature of the mixed solution in it. The temperature of the solution must always be a few degrees above the water temperature so that the heat of condensation and absorption of the refrigerant vapor will flow into the cooling water. Thus, since the pressure is approximately the same as that of the evaporator, the exact condition of the solution may be determined at once. Knowing the temperature and pressure of the chosen refrigerant, the maximum concentration can be determined by the experimental formula or by related tables.

In a similar manner, the pressure in the generator and the rectifier is determined by the condenser pressure, which in turn depends upon the condenser water temperature. Also it is obvious that heat must be supplied to the generator to remove the refrigerant vapor from the mixed solution in the generator. This is generally supplied by saturated steam, and the temperature of the steam must be a few degrees above the temperature of the solution in order to cause the heat to flow into the solution, thereby

distilling off the refrigerant vapor and some absorbent vapor. Hence, since the pressure and temperature are easily determined, the maximum concentration can be found readily.

From the above discussions, it will be noted that the temperature of the cooling water and the temperature that is desired in the cold storage room are principal determining factors.

In accordance with these principal determining factors, and reasonably made assumptions, a typical example for computing the performance of an ammonia absorption system is given in the following section.

3.2. Example of a Typical Ammonia-Absorption Refrigeration System

The following assumptions are made:

The pressures in the generator, the rectifier, the condenser, and the heat exchanger are identical and are equal to 180.6 psia

The temperature of the atmosphere is 540°R

The temperature in the evaporator is 470°R

The vapor leaving the rectifier (1)* is pure NH_3 and is at the saturation temperature corresponding to its pressure.

There is sufficient turbulence in the generator during the mixing of strong and weak solutions; the concentration existing

* The numbers in parentheses refer to the sections correspondingly labeled in Fig. 5.

in the mixed solution is in a complete uniform condition so that each of the streams at sections (8) and (9) has the maximum concentration of ammonia compatible with the pressure and temperature in the generator. Consequently, the weak solution leaving the rectifier, (10), is in equilibrium with the vapor entering it and is therefore of the same concentration and is the same state as the solution leaving the generator (8).

The liquid leaving the condenser (2) is saturated liquid at the pressure of the condenser.

The vapor leaving the evaporator (3) is saturated vapor at the pressure of the evaporator.

The weak solution enters the absorber (4a) at 560°R and leaves (5) at 540°R as a strong solution.

The maximum, or equilibrium concentration at (5) can be approximately evaluated by the Mollier equation[14]

$$x_5 = 2.146 \left(\frac{\overline{T}_e}{\overline{T}_a} - 0.656\right)$$

where $\overline{T}_e$ and $\overline{T}_a$ are average temperatures at the evaporator and the absorber, respectively. Therefore

$$x_5 = 2.146 \left(\frac{470}{540} - 0.656\right)$$

$$= 0.45924$$

Note that this maximum value can also be checked by Fig. 3, or by Fig. 19. Aqua-ammonia Chart of reference (9) (by Emswiller and Schwartz, McGraw-Hill Book Co.), or by Table——Properties of Solutions of Ammonia and Water (page 346 of reference (5) by H.

J. Macintire, John Wiley & Co.). In an ideal case, if we assume that there is a sufficiently large vessel for the absorber with a design mechanism which will cause a complete mixing condition in the absorber, then the concentration of the fluid at the outlet stream of the absorber can reach the above value. However, in an actual design problem, the concentration of the outlet stream of the absorber is always less than this value. Accordingly, assume $x_5 = x_s = 0.29$.

(1) Mass Rate of Flow at Each Section

$$M_1 = M_2 = M_{2a} = M_3 = 1 \quad lb_m \text{ of } NH_3$$

Let

M_7 = pounds of strong solution circulated per pound of ammonia condensed

M_8 = pounds of weak solution circulated per pound of ammonia condensed

x_7 = weight concentration of strong solution = 0.29.

The weak solution leaving the generator (8), $x_8 = 0.13$ by weight which is the maximum concentration of ammonia compatible with the pressure and temperature in the generator.

For steady state conditions,

$$M_7 - M_8 = 1 \tag{63}$$

and

$$M_7 x_7 - M_8 x_8 = 1 . \tag{64}$$

Solving equations (63) and (64), we get

$$M_7 = \frac{1 - x_3}{x_7 - x_8} = \frac{1 - 0.13}{0.29 - 0.13} = 5.438$$

lb_m/lb_m NH_3 condensed

and

$$M_8 = 5.438-1 = 4.438$$

lb_m/lb_m NH_3 condensed

Equating mass flows into and out of the rectifier,

$$M_9 - M_{10} = 1 \qquad (65)$$

By assumption $x_8 = x_{10} = 0.13$. The concentration x_9, on the other hand, is that of vapor which is in equilibrium with a 13 per cent solution of ammonia at the pressure existing in the generator. Employing the chart of the enthalpy-concentration diagram[8] for the system NH_3-H_2O, we get

$$x_9 = 0.602$$

For the ammonia alone, we have

$$x_9M_9 - x_{10}M_{10} = 1 \qquad (66)$$

Solving equations (65) and (66), we get

$$M_9 = \frac{1 - x_{10}}{x_9 - x_{10}}$$

$$= \frac{1 - 0.13}{0.602 - 0.13}$$

$$= 1.843 \qquad lb_m/lb_m \; NH_3 \text{ condensed}$$

and

$$M_{10} = M_9 - M_1$$

$$= 1.843 - 1$$

$$= 0.843 \qquad lb_m/lb_m\ NH_3 \text{ condensed}$$

(2) Energy Balance

Before employing the equations which have been derived in the previous sections to calculate the energy balance, the source data for the calculation will be noted here. Values of availabilities at each section are taken from Warburton* (see page 305 of reference (6)) where the properties of ammonia are from Jennings and Shannon. In order to check Warburton's values of availability, ammonia tables from the United States Bureau of Standards(3) were used. As the base for the ammonia tables of Jennings and Shannon is different than that for the United States Bureau of Standards Tables, a correction for both the enthalpy and entropy terms was necessary, which was done in the following manner: In a footnote to Table IV, page 254 of reference (6), it is stated that the values of enthalpy taken from the United States Bureau of Standards Tables were reduced by 77.9 BTU/lb_m. From Warburton's values of availability at the inlet to the condenser, the inlet and outlet of the expansion valve, and the outlet from the evaporator, the values for entropy were calculated. These entropy values were then compared with those of the United States Bureau of

* T. Warburton, master's thesis, M.I.T., 1938.

Standards Tables. In each case it was found that Warburton's values of entropy were less by 0.1442 BTU/lb_m ^oR than those found in the United States Bureau of Standards Tables.

By means of equations (10), (11), (12), (13), (14), (15), and (16) we get

$$Q_g = M_8h_8 + M_9h_9 - M_7h_7 - M_{10}h_{10}$$

$$= 4.438x239 + 1.843x890 - 5.438x134.1 - 0.843x239$$

$$= 1770.33 \quad BTU/lb_m\ NH_3 \text{ condensed}$$

$$Q_r = h_1 + M_{10}h_{10} - M_9h_9$$

$$= 554.13 + 0.843x239 - 1.843x890$$

$$= -884.84 \quad BTU/lb_m\ NH_3 \text{ condensed}$$

$$Q_c = h_2 - h_1$$

$$= 65.6 - 554.1$$

$$= -448.50 \quad BTU/lb_m\ NH_3 \text{ condensed}$$

$$Q_e = h_3 - h_{2a}$$

$$= 537.0 - 65.6$$

$$= 471.40 \quad BTU/lb_m\ NH_3 \text{ condensed}$$

$$Q_a = M_5h_5 - M_{4a}h_{4a} - h_3$$

$$= 5.438x(-40.5) - 4.438x25.5 - 537.0$$

$$= -870.38 \quad \text{BTU/lb}_m \text{ NH}_3 \text{ condensed}$$

$$Q_p = M_5(h_6 - h_5)$$

$$= 5.438\ (-40.1 + 40.5)$$

$$= 2.18 \quad \text{BTU/lb}_m \text{ NH}_3 \text{ condensed}$$

Or by equation (15)

$$Q_p = \frac{144\ G\ V_S(P_H - P_L)}{778}$$

$$= \frac{144 \text{ x } 5.4375 \text{ x } 0.01794(180.6 - 38.5)}{778}$$

$$= 2.5656 \quad \text{BTU/lb}_m \text{ NH}_3 \text{ condensed}$$

in which the value of V_S, which is a function of temperature and weight concentration, is taken by interpolation from the table of "Specific Volume of Aqua Ammonia Solution"[9] by Jennings and Shannon.

Substituting these values into equation (17), we have

$$Q_g + Q_e + Q_p = 2243.91$$

and

$$Q_a + Q_c + Q_r = -2243.72$$

The right hand side of equation (17) must be zero. The trivial discrepancy occurs from the concerned values of weak and strong solutions which are carried to finite decimal places.

(3) Steam Consumption

The steam consumption at the generator per hour per ton of refrigeration can be evaluated by equation (18), i.e.

$$\text{Steam Consumption} = \frac{Q_g}{Q_e}\frac{12{,}000}{h_{fg}} = \frac{1770.33}{471.40} \cdot \frac{12{,}000}{906.17}$$

$$= 49.73 \quad lb_m \text{ of steam/hr ton.}$$

(4) Performance Ratio[10]

According to the equations derived in Sec. 2.3,

$$Q_{pp} = \frac{T_g}{T_g - T_0} Q_p = \frac{765.3}{765.3 - 540} \times 2.18 = 7.380$$

$$BTU/lb_m \; NH_3 \text{ condensed}$$

$$\Phi_\alpha = \frac{Q_e}{Q_g + Q_{pp}} = \frac{471.40}{1770.33 + 7.38} = 0.2652$$

$$(CP)_{rev} = \frac{T_e}{T_0 - T_e} = \frac{470}{540 - 470} = 6.7143$$

$$(\eta)_{rev} = \frac{T_g - T_0}{T_g} = \frac{765.3 - 540}{765.3} = 0.2944$$

$$\Phi_{\alpha max} = (CP)(\eta) = 6.7143 \times 0.2944 = 1.9767$$

$$\epsilon = \frac{\Phi_\alpha}{\Phi_{\alpha max}} = \frac{0.2652}{1.9767} = 0.134$$

$$\mu = 1 - \epsilon = 1 - 0.134 = 0.866.$$

In comparison with a corresponding compression refrigeration system, assuming the thermal efficiency, η_t, of a power installation, including power transmission losses, is $\eta_t = 0.15$.

Then, by equation (48),

$$(CP)_{\alpha} = \frac{\Phi_{\alpha}}{\eta_t} = \frac{0.2652}{0.15} = 1.7686$$

This states that an absorption refrigeration cycle, which has a value of $\Phi_{\alpha} \geq 0.2652$, can compete with a compression refrigeration cycle with a coefficient of performance, $(CP)_k = 1.7686$.

Figure 6 shows a diagram for the ideal, or reversible, absorption refrigeration cycle which operates at the same temperature levels and the refrigerating load, $Q_e = 471.4$ BTU/lb_m NH_3 condensed, as the numerical problem discussed in this section. Area (D+E) represents the refrigerating load, Q_e. The ideal work required for operating the refrigerator is

$$W_{ideal} = Q_g \frac{T_g - T_0}{T_g} = Q_e \frac{T_e - T_0}{T_e}$$

which is represented by the area A, or (B+C). Area (A+B+D) represents the ideal amount of heat supplied, Q_g. The ideal coefficient of performance is thus evaluated by

$$(CP)_{ideal} = \frac{Q_e}{Q_g} = \frac{T_e}{T_g} \frac{T_g - T_0}{T_0 - T_e} = \frac{471.4}{238.48} = \frac{470}{765.3} \frac{765.3 - 540}{540 - 470}$$

$$= 1.98.$$

(5) Entropy Change in Each Apparatus

The amount of entropy change in each apparatus, in the actual case, can be calculated by use of the following formula for the difference in entropies of the exit and in-

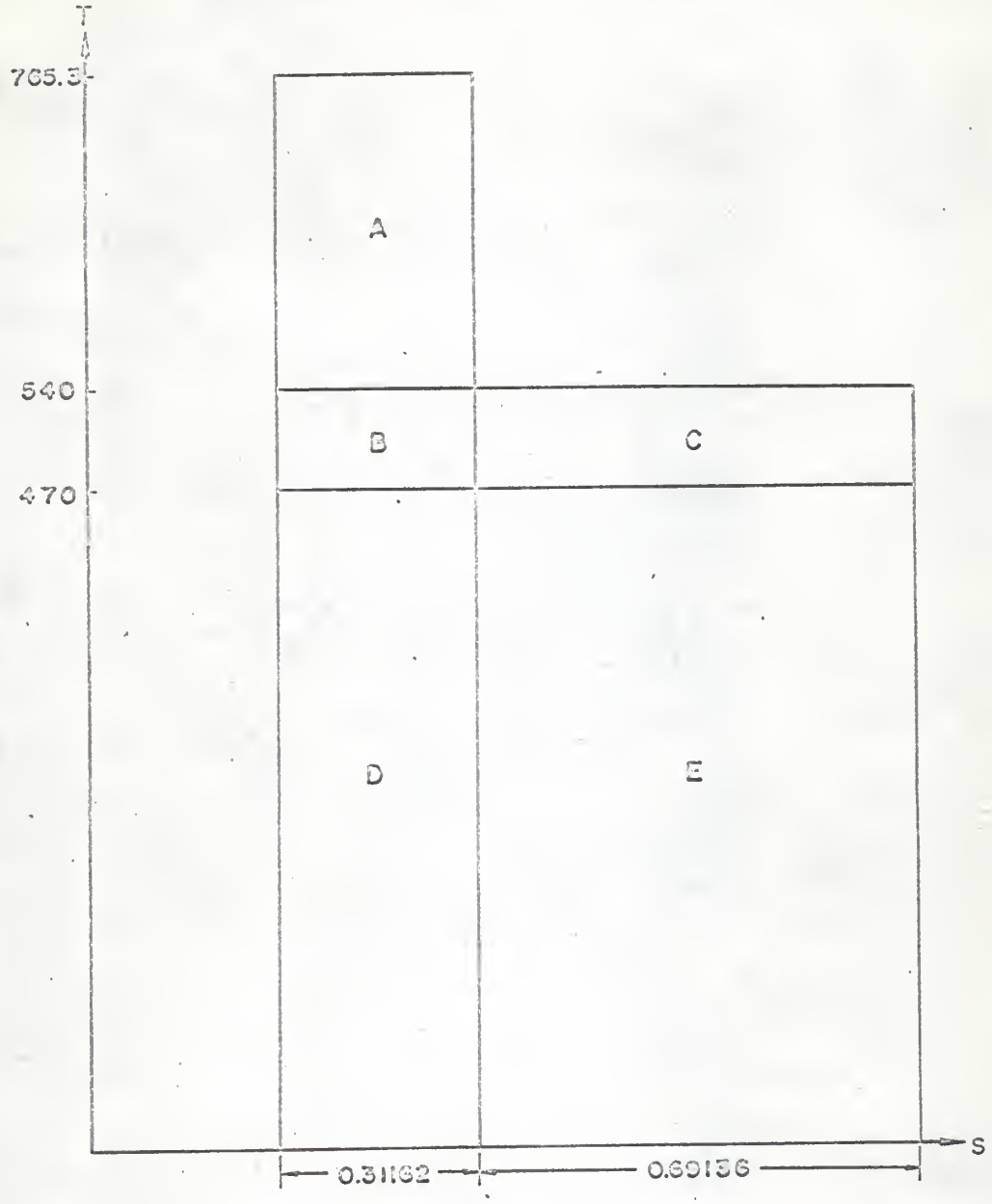

Fig. 6. Diagram for an ideal absorption refrigeration cycle

let fluid streams,

$$\Delta S_{act} = \Sigma(MS)_{out} - \Sigma(MS)_{in}$$

Referring to equations derived in Sec. 2.3, we can calculate the entropy change in each apparatus.

Furthermore, the creation of entropy of the ammonia, due to friction, is zero in the rectifier condenser, the main condenser and the evaporator, as these processes were assumed to be reversible. Likewise the creation of the entropy of the cooling water, due to friction, is zero in the rectifier condenser, the main condenser and the absorber. However, in the rectifier condenser and the main condenser heat is transferred from the ammonia at 550°R to the cooling water at 540°R. These heat transfers through a finite temperature difference of 10°F cause creations of entropy as follows:

$$\Delta S_{act,a} = M_5 S_5 - M_{4a} S_{4a} - S_3$$

$$= 5.438 \times 0.0548 - 4.438 \times 0.1270 - 1.1715$$

$$= -1.437 \quad \text{BTU/°R lb}_m \text{ NH}_3 \text{ condensed}$$

$$S_{created,a} = \Delta S_{act} - \frac{Q_a}{T_0}$$

$$= -1.4371 - \frac{(-870.38)}{540}$$

$$= 0.175 \quad \text{BTU/°R lb}_m \text{ NH}_3 \text{ condensed}$$

$$\Delta S_{act,c} = S_2 - S_1$$

$$= 0.1515 - 1.0404$$

$$= -0.889 \quad \text{BTU/}^{\circ}\text{R lb}_m \text{ NH}_3 \text{ condensed}$$

$$S_{created,c,\Delta T} = Q_c \left(\frac{1}{T_0} - \frac{1}{T_c}\right)$$

$$= 488.5 \left(\frac{1}{540} - \frac{1}{550}\right)$$

$$= 0.016 \quad \text{BTU/}^{\circ}\text{R lb}_m \text{ NH}_3 \text{ condensed}$$

$$\Delta S_{act,r} = S_1 + M_{10}S_{10} - M_9S_9$$

$$= 1.0404 + 0.843\text{x}0.4398 - 1.843\text{x}1.4272$$

$$= -1.219 \quad \text{BTU/}^{\circ}\text{R lb}_m \text{ NH}_3 \text{ condensed}$$

$$S_{created,r} = \Delta S_{act,r} - \frac{Q_r}{T_r}$$

$$= -1.2193 + 1.6088$$

$$= 0.390 \quad \text{BTU/}^{\circ}\text{R lb}_m \text{ NH}_3 \text{ condensed}$$

The entropy creation, due to the finite temperature in the rectifier, is

$$S_{created,r,\Delta T} = Q_r \left(\frac{1}{T_0} - \frac{1}{T_r}\right)$$

$$= 884.84 \left(\frac{1}{540} - \frac{1}{550}\right)$$

$$= 0.030 \quad \text{BTU/}^{\circ}\text{R lb}_m \text{ NH}_3 \text{ condensed}$$

Therefore, the total creation of entropy in the rectifier is 0.420 BTU/$^{\circ}$R lb$_m$ NH$_3$ condensed.

$$\Delta S_{act,e} = S_3 - S_{2a} = 1.1715 - 0.1674 = 1.004$$

$$S_{created,e} = 0 \quad \text{BTU/}^{o}\text{R lb}_m \text{ NH}_3$$

$$\Delta S_{act,g} = M_8S_8 + M_9S_9 - M_7S_7 - M_{10}S_{10}$$

$$= 4.438x0.4398 + 1.843x1.4272$$

$$-5.438x0.3396 - 0.843x0.4398$$

$$= 2.365 \quad \text{BTU/}^{o}\text{R lb}_m \text{ NH}_3 \text{ condensed}$$

$$S_{created,g} = S_{act,g} - \frac{Q_g}{T_g} = 2.3647 - \frac{1770.33}{765.3}$$

$$= 0.051 \quad \text{BTU/}^{o}\text{R lb}_m \text{ NH}_3 \text{ condensed}$$

$$\Delta S_{act,ex} = S_{created,ex} = M_6(S_7 - S_6) + M_4(S_4 - S_8)$$

$$= 5.438(0.3396-0.0548) + 4.438(0.1263-0.4398)$$

$$= 0.157 \quad \text{BTU/}^{o}\text{R lb}_m \text{ NH}_3 \text{ condensed}$$

$$\Delta S_{act,vs} = S_{created,vs} = M_4(S_{4a} - S_4)$$

$$= 4.438 (0.1270 - 0.1263)$$

$$= 0.003 \quad \text{BTU/}^{o}\text{R lb}_m \text{ NH}_3 \text{ condensed}$$

$$\Delta S_{act,ve} = S_{created,ve} = S_{2a} - S_2$$

$$= 0.1674 - 0.1515$$

$$= 0.0160 \quad \text{BTU/}^{o}\text{R lb}_m \text{ NH}_3 \text{ condensed}$$

$$\Delta S_{act,p} = S_{created,p} = 0$$

$$\Sigma S_{created} = S_{created,a} + S_{created,r,\Delta T} + S_{created,g} + S_{created,c,\Delta T} + S_{created,ex} + S_{created,vs} + S_{created,ve}$$

$$= 0.8084 \quad BTU/^{o}R\ lb_m\ NH_3\ condensed$$

$$\Delta S_{atmosphere} = (S_e + S_g) + \Sigma S_{created}$$

$$= 3.3688 + 0.838$$

$$= 4.2068 \quad BTU/^{o}R\ lb_m\ NH_3\ condensed$$

(6) Changes of Availabilities and Irreversibilities

By equations (59) and (61), changes of availability and irreversibility for each process can be evaluated as follows:

$$\Delta B_a = M_5 b_5 - M_{4a} b_{4a} - b_3$$

$$= 5.438x(-70.1) - 4.438x(-43.1) - (-95.6)$$

$$= -94.31 \quad BTU/lb_m\ NH_3\ condensed$$

$$I_a = -\Delta B_a$$

$$= 94.31 \quad BTU/lb_m\ NH_3\ condensed$$

$$\Delta B_c = b_2 - b_1$$

$$= -16.2 - (-7.7)$$

$$= -8.5 \quad BTU/lb_m\ NH_3\ condensed$$

Further

$$\Delta B_c = b_2 - b_1$$

$$= (h_2 - T_0 S_2) - (h_1 - T_0 S_1)$$

$$= h_{fg} - T_0 \frac{Q_c}{T_c}$$

$$= Q_c \frac{T_c - T_0}{T_c}$$

Therefore, $I_c = 0$ for ammonia only. This can be seen by referring to equation (61). However, due to the finite temperature difference when heat is transferred from the ammonia to the cooling water, the available energy degraded is

$$I'_c = T_0 \left[Q_c \left(\frac{1}{T_0} - \frac{1}{T_c}\right)\right]$$

$$= 540 \left[488.5 \left(\frac{1}{540} - \frac{1}{550}\right)\right]$$

$$= 8.86 \quad \text{BTU/lb}_m \text{ NH}_3 \text{ condensed}$$

$$\Delta B_r = b_1 + M_{10} b_{10} - M_9 b_9$$

$$= -7.7 + 0.843 \times 1.5 - 1.843 \times 119.3$$

$$= -226.33 \quad \text{BTU/lb}_m \text{ NH}_3 \text{ condensed}$$

$$I_r = Q_r \frac{T_r - T_0}{T_r} - \Delta B_r$$

$$= (-884.8) \frac{550 - 540}{550} - (-226.33)$$

$$= 210.24 \quad \text{BTU/lb}_m \text{ NH}_3 \text{ condensed}$$

The irreversibility created by heat transfer through a finite temperature difference can be evaluated by

$$I'_r = T_0 \left[Q_r \left(\frac{1}{T_0} - \frac{1}{T_r} \right) \right] = 540 \left[884.8362 \left(\frac{1}{540} - \frac{1}{550} \right) \right] = 16.09$$

BTU/lb$_m$ NH_3 condensed.

$$\Delta B_e = b_3 - b_2 = -95.6 - (-24.8)$$

$$= -70.80 \quad \text{BTU/lb}_m\ NH_3 \text{ condensed}$$

$$I_e = 0$$

$$\Delta B_g = M_8 b_8 + M_9 b_9 - M_7 b_7 - M_{10} b_{10}$$

$$= 4.438 \times 1.5 + 1.843 \times 119.3 - 5.438 \times (-49.3) - 0.843 \times 1.5$$

$$= 493.36 \quad \text{BTU/lb}_m\ NH_3 \text{ condensed}$$

$$I_g = Q_g \frac{T_g - T_0}{T_g} - \Delta B_g$$

$$= 1770.33 \frac{765.3 - 540}{765.3} - 493.36$$

BTU/lb$_m$ NH_3 condensed

$$= 27.83 \quad \text{BTU/lb}_m\ NH_3 \text{ condensed}$$

$$\Delta B_{ex} = M_7(b_8 - b_6) + M_4(b_4 - b_8)$$

$$= 5.438(-49.3 + 69.7) + 4.438(-42.7 - 1.5)$$

$$= -85.22 \quad \text{BTU/lb}_m\ NH_3 \text{ condensed}$$

$$I_{ex} = - \Delta B_{ex}$$

$$= 85.22 \quad \text{BTU/lb}_m\ NH_3 \text{ condensed}$$

$$\Delta B_{vs} = M_{4a}(b_{4a} - b_4)$$
$$= 4.438(-43.1 + 42.7)$$
$$= -1.78 \quad \text{BTU/lb}_m\ NH_3 \text{ condensed}$$

$$I_{vs} = -\Delta B_{vs}$$
$$= 1.78 \quad \text{BTU/lb}_m\ NH_3 \text{ condensed}$$

$$\Delta B_{ve} = b_{2a} - b_2$$
$$= -24.8 - (-16.2)$$
$$= -8.6 \quad \text{BTU/lb}_m\ NH_3 \text{ condensed}$$

$$I_{ve} = -\Delta B_{ve}$$
$$= 8.6 \quad \text{BTU/lb}_m\ NH_3 \text{ condensed}$$

$$\Delta B_p = -W_p$$
$$= -2.18 \quad \text{BTU/lb}_m\ NH_3 \text{ condensed}$$

$$I_p = 0$$

Section	Phase	M (lb_m)	T (^oR)	P (psia)	h (BTU/lb_m)	S (BTU/$lb_m\,^oR$)	b (BTU/lb_m)
(1)	g	1.0	550.0	180.6	554.1	1.0404	- 7.7
(2)	f	1.0	550.0	180.6	65.6	0.1515	-16.2
(2a)	g+f	1.0	470.0	38.5	65.6	0.1674	-24.8
(3)	g	1.0	470.0	38.5	537.0	1.1715	-95.6
(4)	f	4.438	560.0	180.6	25.5	0.1268	-42.7
(4a)	f	4.438	560.0	38.5	25.5	0.1270	-43.1
(5)	f	5.438	540.0	38.5	-40.5	0.0548	-70.1
(6)	f	5.438	540.0	180.6	-40.1	0.0548	-69.7
(7)	f	5.438	696.0	180.6	134.1	0.3396	-49.3
(8)	f	4.438	765.3	180.6	239.0	0.4398	1.5
(9)	g	1.843	765.3	180.6	890.0	1.4272	119.3
(10)	f	0.843	765.3	180.6	239.0	0.4398	1.5

Table I. State properties of the system calculated in Sec. 2.3.

Apparatus	Heat Received (BTU)	Work Input (BTU)	$\frac{T-T_0}{T} Q$ (BTU)	$T_0 \frac{Q}{T}$ (BTU)	$S_{act.}$ (BTU/^oR lb_m NH_3 condensed)	$S_{created}$ (BTU/^oR lb_m NH_3 condensed)	$T_0 x S_{created}$ (BTU)
Absorber	-870.38	0.0	0	-870.38	-1.437	0.175	94.338
Condenser	-488.50	0.0	- 8.88	-479.62	-0.889	0.016	8.856
Rectifier	-884.84	0.0	-16.09	-868.75	-1.219	0.42	226.422
Evaporator	471.40	0.0	-70.21	541.61	1.004	0.0	0.0
Generator	1770.33	0.0	521.19	1249.14	2.365	0.051	27.756
Heat Exchanger	0.0	0.0	------	-------	0.157	0.157	84.996
Solution Valve	0.0	0.0	------	-------	0.003	0.003	1.674
Expansion Valve	0.0	0.0	------	-------	0.016	0.016	8.586
Pump	0.0	2.18	------	-------	0.0	0.0	0.0
Summation	-1.99	2.18	426.01	-428.00	0.00	0.838	452.628

Table II. Entropy flow and entropy changes in each apparatus (refer to Sec. 3.2.)

Apparatus	ΔB_- (BTU/ lb_m NH_3 condensed)	ΔB_-(%)	ΔB_+ (BTU/ lb_m NH_3 condensed)	ΔB_+(%)	I (BTU/ lb_m NH_3 condensed)	I(%)	$^{BTU}/_{lb_m NH_3}$ I' (condensed)	I'(%)
Absorber	94.30	19.03	------	-----	94.31	22.04	94.31	20.82
Condenser	8.50	1.71	------	-----	0.0	-----	8.86	1.96
Rectifier	226.33	45.67	------	-----	210.24	49.13	226.34	49.97
Evaporator	70.80	14.29	------	-----	0.0	0.0	0.0	0.00
Generator	-----	-----	493.36	99.56	27.83	6.50	27.83	6.14
Heat Exchanger	85.21	17.20	------	-----	85.21	19.91	85.21	18.81
Solution Valve	1.78	0.36	------	-----	1.78	0.41	1.78	0.39
Expansion Valve	8.60	1.74	------	-----	8.60	2.01	8.60	1.90
Pump	-----	-----	2.18	0.44	0.0	-----	0.0	-----
Summation	495.52	100.0	495.54	100.0	427.97	100.0	452.23	100.00

Table III. Changes of availabilities and irreversibilities through each apparatus (refer to Sec. 3.2).

3.3. Conclusions

In relation to the previous discussions, there are some important conclusions which will be shown as follows:

(1) For the cycle, the summation of the quantities of the net heat received and the net work input must be zero. This can also be expressed by the equational form[16] with numerical values taken from the example in Sec. 3.2.

$$Q_{in} + W_{in} = Q_0$$
$$2241.73 + 2.57 = 2244.30 \doteq 2243.72$$

in which $Q_{in} = Q_e + Q_g$, $W_{in} = Q_p$ and $Q_0 = Q_a + Q_c + Q_r$.

(2) In each process in which there is heat flow, the available and unavailable parts of the heat flow equals the total heat flow. For the total of the evaporator and the generator

$$Q_{in} = (AE)_{Q_{in}} \quad (UE)_{Q_{in}}$$
$$2241.73 = 450.98 + 1790.75$$

and for the total of the condenser, absorber and rectifier,

$$Q_0 = (AE)_{Q_0} + (UE)_{Q_0}$$
$$2243.71 = 24.97 + 2218.74$$

(3) The net change of entropy for any cycle, ΔS_{act}, must be zero.

(4) For the working fluid in cycle, the summation of the total increase in availability, $\Sigma\Delta B_+$, and the total decrease in availability, $\Sigma\Delta B_-$, must be zero.

(5) All rotary shaft work, such as pump work, is available.

(6) From Table III, we can see that the greatest increase in the irreversibility occurs in the rectifier. This is mainly due to the mixing of fluids of large temperature difference.

(7) The next largest increase in the irreversibility occurs in the absorber. This is principally due to the mixing of fluids of different temperatures (the temperature difference of mixing fluids in the absorber is less than those in the rectifier).

(8) For the cycle,

$$\Delta S_{universe} = \Sigma S_{created} = \Delta S_{atmosphere} - (S_e + S_g)$$

$$0.838 = 0.838 = 4.206 - 3.368$$

(9) The actual amount of work, which is required to operate the unit, can be evaluated by

$$W_{in,act} = W_{in,ideal} + T_0 \Delta S_{universe}$$

$$523.26 \doteq 70.21 + 452.63 = 522.84$$

(10) The increase in unavailable energy, caused by the system alone as it completes one cycle of operation can also be evaluated by $-T_0 \Sigma \frac{Q}{T}$. The total increase in unavailability, due to the above term and the increase caused by transfer of heat through a finite temperature difference to the cooling water in the condenser and the rectifier, can be evaluated by $T_0 \Delta S_{universe}$.

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NOMENCLATURE

(AE)	Available part of internal energy	BTU
b	Availability per unit mass	BTU/lb_m
B	Available part of enthalpy	BTU
(CP)	Coefficient of performance	dimensionless
E	Energy	BTU
f	Liquid state	dimensionless
g	Vapor state	dimensionless
G	Pounds of strong solution circulated per pound of refrigerant	dimensionless
h	Enthalpy	BTU/lb_m
I	Irreversibility	BTU
I'	(Quantity of available energy degrades to the cooling water) ÷ I	BTU
J	Conversion factor, 778	$ft - lb_f/BTU$
M	Mass	lb_m
P	Pressure	psia
q	Quantity of heat transferred per unit mass	BTU/lb_m
Q	Quantity of heat transfer	BTU
s	Entropy	$BTU/^oR\ lb_m$
T	Absolute temperature	^oR
u	Internal energy per unit mass	BTU/lb_m
(UE)	Unavailable energy	BTU
v	Specific volume	ft^3/lb_m
W	Work	$ft - lb_f$
x	Weight concentration	lb_m/lb_m

Φ	Performance ratio	dimensionless
η	Thermal efficiency	dimensionless

Subscripts

a	Absorber
act.	Actual conditions
c	Condenser
D	Dead state
e	Evaporator
ex	Heat exchanger
f	Friction loss
g	Generator
H	High
irr	Irreversible process
k	Compression refrigeration cycle
L	Low
max.	Maximum value
p	Pump
pp	Defined as equation (22) on page 25
r	Rectifier
rev	Reversible process
s	Strong solution
t	Thermal property
ve	Expansion valve
vs	Solution valve
w	Weak solution

α	Absorption refrigeration cycle
0	Conditions of surroundings
+	Increment
-	Decrement

ABSORPTION REFRIGERATION SYSTEM, 1965-1966

by

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B.S., Taiwan Provincial Cheng Kung University, 1958

AN ABSTRACT OF A MASTER'S REPORT

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The field of refrigeration is a broad subject, and involves the application of many basic studies. In this report the absorption refrigeration system has been analyzed from the thermodynamic viewpoint.

The report consists of three sections. In the first section general concepts of absorption refrigeration systems are outlined, starting with a comparison with a simpler system ---- the compression refrigeration system. In the second section the absorption refrigeration system is analyzed by the First-Law and Second-Law of thermodynamics, involving mass balance, energy balance, steam consumption, entropy change, performance ratio, and the concepts of availability and irreversibility. In the last section a numerical example of a typical ammonia absorption refrigeration system is presented, employing the described principles and derived formulas of previous sections. Availabilities and irreversibilities are tabulated by percentages on Table III. The rectifier accounts for approximately one-half of the total irreversibility of the system, while the absorber and the heat exchanger each accounts for about one-fifth of the total degradation of energy. Effective changes in the design of these three pieces of equipment would materially improve the performance of the system.